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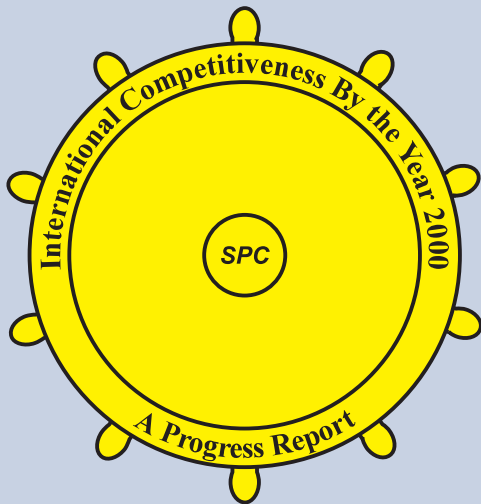
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Development Of A Production Optimization Program For Design And Manufacture Of Light Weight/High Strength Hull For The Next Generation Of High Speed Craft

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ABSTRACT

There is an interest in introducing a high speed marine vehicle for crew boat service to the offshore oil and gas fields in the Gulf of Mexico. Consequently, it is necessary to develop a light weight hull structure suitable for rapid modular construction. This paper presents the authors' numerical and experimental evaluation of a lightweight aluminum hull panel. An optimization routine was developed to investigate the sensitivity of the design to different structural arrangements. An example of the optimization routine for a stiffened aluminum plate is presented.

INTRODUCTION

The recent increase in oil prices has created a resurgence in oil and gas field development. These new fields are farther offshore and in deeper water. This development is impacting both rig construction as well as field support vessels such as crew boats and offshore supply boats.

Traditionally, crew exchange has been done using helicopters. However the deep water fields are often outside the helicopter's operating range. The helicopters have had to land on near-shore platforms and re-fuel to reach the new offshore fields. These offshore fields are also creating service requirements which are difficult for the helicopter to meet due to their limitations from weather, payload, and fuel capacity.

This situation has opened the possibility of introducing a 30 - 42 knot crew boat for this deep water offshore crew/cargo exchange. This new generation of crew boat can be built in a cost effective manner by taking advantage of advances in ship production technology, especially in the areas of engineering design and manufacturing.

In order to properly develop this high speed crew boat, it is necessary to develop the craft in all four quadrants of the technology cross [1]:
1. Materials,
2. Structure/Construction,
3. Propulsion System, and
4. Hull form - Resistance and seakeeping.

This paper discusses an ongoing research project that focuses on quadrant 2, Structure/Construction. This work is part of a two year research project sponsored by the Gulf Coast Region Maritime Technology Center (GCRMTC).

SERVICE REQUIREMENTS

There has been a gradual evolution in the design of offshore crew boat vessels [2]. With the development of the deep water offshore fields in the Gulf of Mexico, it becomes difficult to make crew changes exclusively by helicopter. Therefore an emerging requirement exists for a 40 - 45 m long, 30 - 42 knot crew boat, capable of meeting the requirements outlined in Table I.

Today a number of 35 - 40 knot high speed aluminum catamarans are operating worldwide [3]. They have become a reliable high speed passenger and cargo transport craft. A catamaran vessel, with its large deck area, is also attractive for offshore crew boat service. At 35 - 40 knots, the crew transfer could be within an acceptable 2 - 3 hour duration.

Vessel Speed	30 - 40 kts
Vessel Cargo	50 - 100 tons max
Vessel Range	500 - 600 miles
Passengers	10 - 12

Table I. Next generation crew boat high speed cargo vessel requirements.

VESSEL DESIGN

The preliminary design of the vessel resulted in the principal particulars listed in Table II.

Catamaran	Units	Value
Length	m (ft)	40 (125)
Beam (overall)	m (ft)	10.5 (34.5)
Beam (hull)	m (ft)	2.743 (9)
Draft	m (ft)	1 (3.33)
Displacement	tons	120 - 150
Speed	knots	35
Material	Aluminum	
Engine	Diesel	

Table II. Vessel Particulars

The half midship section arrangement is shown in Figure 1. The hull form is a surface piercing type. It is to be manufactured in modules which are assembled in a panel line. Since the material flow is critical, the panels would be manufactured from aluminum plate and readily available structural extrusions.
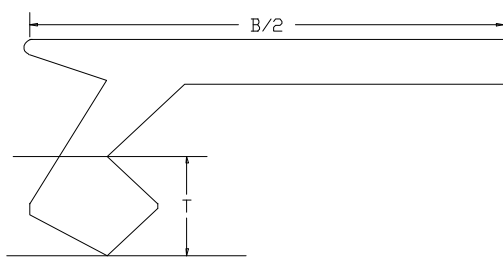

Figure 1. Surface piercing hull form.

HULL PANEL DEVELOPMENT

The hull structure was developed to satisfy three requirements:
1. Classification Rules
   To make this vessel marketable worldwide, it is necessary to satisfy classification society rules such as DNV, Bureau Veritas, as well as the new ABS rules for high speed craft [4,5].
2. Modular Construction
   The hull structure was designed to be manufactured from aluminum stock plate and readily available aluminum structural extrusions. This is reflected in the hull panel geometry summarized in Table III.
3. Floating Frame Arrangement
   The third aspect of the structural design is to incorporate the floating frame. The floating transverse frame is welded on the upper flange of the longitudinals. It offers a reduction in welding man-hours and fit-up at some loss of panel stiffness.

The resulting panel is shown in Figure 2. The details of the panel geometry are summarized in Table III.
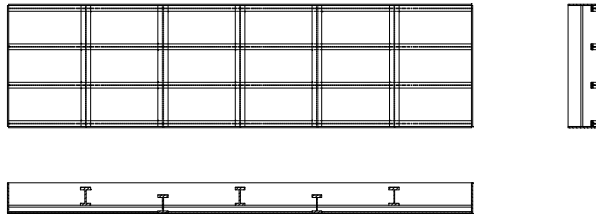

Figure 2. Floating frame hull panel.

Item	Value/Description
Material	Aluminum
Length	4.572 m (15 ft)
Width	1.829 m (6 ft)
Plate thickness	.794 cm (.3125 in)
Longitudinal stiffeners	7.62 cm (3 in) Al I-beam
Transverse stiffeners	17.78 cm (7 in) Al I-beam

Table III. Hull Panel Geometry

DESIGN LOADS AND TEST PANEL DESIGN

A comparison of the various applicable classification rules indicated a similarity in the hull design pressures [6]. A large number of Australian built passenger catamarans are classed using the DNV rules. The test panel design was checked using the DNV rules. As shown in Table IV, the proposed panel geometry, thickness, and structural allowables satisfies the appropriate DNV rules.

DEVELOPMENT OF PREDICTIVE COMPUTER ANALYSIS OF HULL STRUCTURE PANEL

The problem addressed was how the panel could be

DNV Rule [4]	Item	Required	Actual
5. B 101	Plating thickness	6.19 mm (0.244 in)	7.94 mm (0.3125 in)
5. B 202	Plating thickness	6.22 mm (0.245 in)	7.94 mm (0.3125 in)
5. B 302	Plating thickness	5.28 mm (0.208 in)	7.94 mm (0.3125 in)
5. C 101	Long. stiffener section modulus	24.9 cm^3 (1.52 in^3)	27.5 cm^3 (1.68 in^3)
5. C 201	Long. stiffener section modulus	18.1 cm^3 (1.104 in^3)	27.5 cm^3 (1.68 in^3)

Table IV. DNV rule check of panel design.

designed to have adequate strength and minimum cost. The cost savings would be realized in terms of:
   1) Reduction in material and welding,
   2) Reduction in hull weight,
   3) Reduction in production man-hours.
To address this problem, a joint university-industry research project was initiated under the support of GCRMTC. This study is in three parts:
Part I   Design of aluminum test panel,
Part II   FEM analysis of test panel and comparison with   Part III results to improve predictive load,   elongation, and stress prediction capability,
Part III   Manufacture of structural test system and   physical tests of aluminum test panel.
Parts I, II, and III were performed concurrently. For example, the panel design was developed in conjunction with the design of the structural tester [7].

The test panel was sized to enable a valid comparison of the present results with the FE analyses. Earlier tests performed by

2

Clarkson [8] using 3 ft x 3 ft and 4 ft x 3 ft steel hull panel grillages showed the section of a 5 ft x 15 ft panel would be more than adequate for the present analysis. This opens the possibility of studying both the structural response as well as the fatigue strength of the welds.

Physical testing of the panel was performed using a structural test system. Here the test panel is mounted within the structural test system as shown in Figure 3. Multiple hydraulic actuators are used to simulate the design pressure loading. Load, strain, and deflection measurements are recorded at various locations on the panel. Table V summarizes the location of load, deflection, and strain data recorded.

The actuator loads were applied slowly up to a total of 6000 lbs. Repeated tests showed a maximum deflection of 0.071 inch at this 6000 lb loading. This compares well with the 0.084 inch mid-area deflection
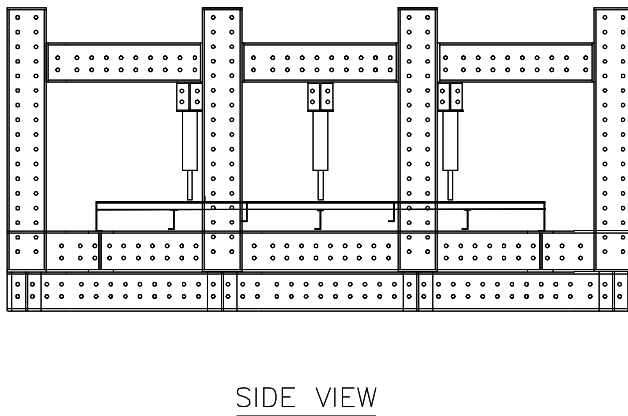

SIDE VIEW

Figure 3. Panel in Test Frame

Quantity	Measurement	Location
2	Deflection	Center longitudinal
1	Deflection	Center fixed transverse
1	Deflection	Center floating transverse
2	Strain (rosette)	Shell plating
4	Strain gage	Center longitudinal
1	Strain gage	Center fixed transverse
1	Strain gage	Center floating transverse

Table V. Location of deflection and strain gages.

predicted by the finite element model. The small difference in results may be due to several factors: 1) boundary conditions around the panel edges, not acting as knife edge supports, 2) differences in the test panel geometry, and 3) thickness variations between the computer model and the test specimen.

Strain gage data were continuously recorded during the loading cycle and an additional test was performed to check for repeatability of results. The applied load and resulting strain for these tests are shown in Figure 4, along with the corresponding finite element predictions. The strain data shown is the average longitudinal strain as read from four gages. Differences between predicted and experimental results are due to a combination of factors. These include differences in actual and modeled boundary

conditions, material and geometry imperfections, and model discretization.

The tests showed the validity of the finite element results in predicting the elastic load response of the panel with floating frames. This provided the basis for the optimization study and follow-on tests with a uniform pressure loading. The uniform pressure loading will be performed by evacuating the panel back using a
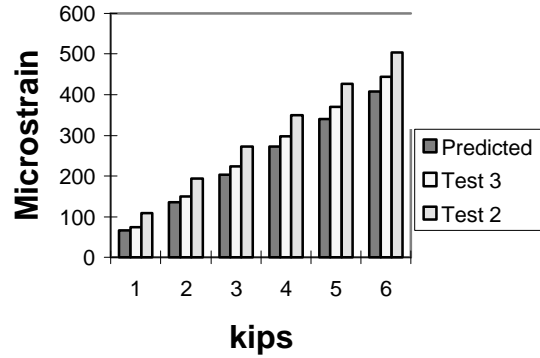

Figure 4. Comparison of Test Data and Finite Element Prediction.

vacuum pump giving,

$$P_{load} = P_{back} - 14.7 \text{ psi.} \qquad (1)$$

These results with the uniform test pressure will be compared to the equivalent loads obtained with the test frame actuators.

FINITE ELEMENT ANALYSIS OF THE TEST PANEL

Finite element analysis of the stiffened plate was performed using the ANSYS® general purpose finite element code. To model the base plate, ANSYS Shell63 quadrilateral elements were used. This element has both bending and membrane capabilities along with six degrees of freedom at each node namely, Ux, Uy, Uz, θx, θy, and θz. The element Beam44, a three dimensional elastic beam element, was used to model the longitudinal and transverse stiffeners. This element also has three translational and three rotational degrees of freedom. A total of 1464 elements were used to model the plated structure. Progressively finer meshes were evaluated until the results converged.

Results of the finite element analyses are shown in Figure 5. Boundary conditions for the analysis were simply supported for the two longitudinal edges which represent longitudinal girders and fixed conditions along the transverse edges to represent transverse bulkheads. Figure 5 is a plot of the out-of-plane displacement field w, for the stiffened panel resisting uniform pressures of 69 KPa (10 psi) and 103 KPa (15 psi). As shown in the
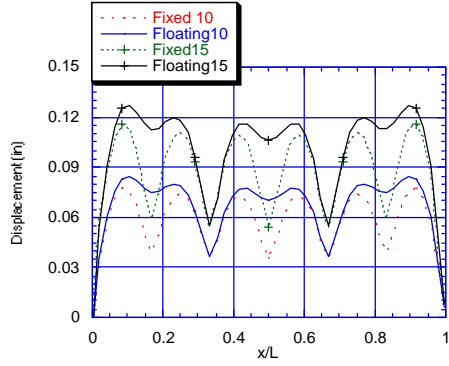
Figure 5. Finite element results of uniform pressure loading.

figure, the maximum displacement occurs with the floating frame system, and has a value of 3.35 mm (0.132 inch).

PANEL OPTIMIZATION

From the standpoint of ship production, it is important before production planning to insure that the items can be produced effectively with minimum cost materials. For high speed craft, the minimization of as-built weight is critical to achieving good performance. This can be accomplished by re-examining the structure and performing an optimization. The optimization was performed using the optimization scheme, HULLOPT.

An optimization scheme, HULLOPT, has been developed to focus on the design of lightweight/high strength hull panels. These stiffened panels will be used in the modular construction of the next generation high speed crew boats. The purpose of HULLOPT is to examine the stiffened panel behavior with different structural elements and panel thickness, in order to determine an optimum structure.

Optimization of the stiffened panel is formulated as a mathematical optimization problem. This is generally written as,

$$\text{minimize: } z = f(X) \qquad (2)$$

$$\text{subject to: } g_i(x) \le \overline{g}_i \qquad i = 1, \ldots, m \qquad (3)$$

$$x_j^{LB} \le x_j \le x_j^{UB} \qquad j = 1, \ldots, n \, (4)$$

where f(X) is the objective function to be minimized, $g_i(x)$ are the m constraints, along with their limits, $\overline{g}_i$. The set of n design variables are given by x_j, with the lower and upper bounds of the design variables given by x_j^{LB}, and x_j^{UB}, respectively.

The objective function for the case of stiffened panels could be to minimize weight, material and labor costs, or a combination of the two. Such a combination would consider minimizing weight in order to increase the load carrying capacity of the vessel, and hence offset greater cargo capacity with initial higher construction costs. In the sample problem solved in this paper, weight is the critical factor in this design, therefore minimum panel

weight is the chosen objective function.

Behavioral inequality constraints are represented in the formulation. These constraints provide limitations on behavioral quantities such as stresses and displacements. In the sample problem that follows, the constraints follow the DNV code for aluminum high speed vessels. These constraints include:

1. minimum plating thickness,
2. minimum section modulus for longitudinal and transverse stiffeners,
3. minimum shear area for longitudinal and transverse stiffeners,
4. maximum allowable buckling stress to prevent web and flange buckling, and
5. maximum allowable local and bending von Mises equivalent stresses for plating and stiffeners.

Two additional geometric constraints were imposed on the optimization problem. The first geometric constraint is that there must be equal spacing between longitudinal and transverse stiffeners. The second constraint requires that the transverse frames alternate between fixed and floating members.

Design variables are the quantities to be determined during an optimization routine. Design variables may be dependent or independent variables that describe the problem to be optimized. For the stiffened plate, six independent design variables are used; plating thickness, longitudinal section modulus, fixed frame section modulus, floating frame section modulus, longitudinal stiffener spacing, and transverse stiffener spacing.

Input to the optimization is the initial panel geometry, thickness and stiffener size. For this analysis, overall plate geometry in terms of length and width, remained constant. Figure 2 shows the stiffened plate with alternating "floating" transverse frames. Plate geometry is given in Table III. The initial design featured a plate thickness of .794 cm (.3125 in), four 7.62 cm (3" x 1.96 lb/ft) extruded Al I-beams for longitudinal stiffeners, and five 17.78 cm (7" x 5.8 lb/ft) extruded Al I-beams for transverse stiffeners. Equal stiffener spacing was used throughout the plate, with a longitudinal spacing of .3048 m (12 in), and a transverse frame spacing of .762 m (30 in). The weight of this panel is 347 kg (765 lb).

In order to determine the sensitivity of the objective function to the design variables, the gradient of the objective function was calculated at the optimum design point. Figure 6 shows the change in objective function versus a plus or minus 1% change in the design variables. In this figure, 'Thick' refers to the plating thickness, 'Iyyt' and 'Iyyl' refer to the moment of inertia in the transverse and longitudinal directions, respectively. As can be seen from the figure, the thickness design variable has the greatest effect on the objective function.
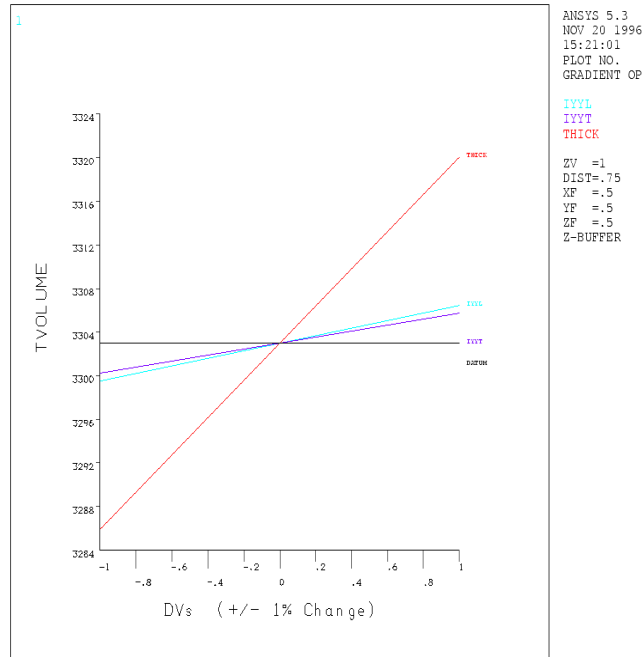
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Figure 6. Gradient of design variables.

The optimization procedure was performed and results were obtained using continuous design variables. However, due to the expense of using non-standard sizes for plating and stiffeners, the optimum sizes were increased to the nearest standard size. Results from the optimization analysis are given in Table VI. In this case, the optimized design varies from the original design in terms of plating thickness, longitudinal stiffener size and spacing, and transverse stiffener size. The final design features a rolled plating thickness of .635 cm (.25 in), which is a standard size. This thinner plating required the use of an additional, yet slightly smaller, longitudinal stiffener. The longitudinal stiffener requirement may be met by the use of five Aluminum Association extruded standard I-beams 7.62 cm (3" x 1.64 lb/ft), with a section modulus of 24.42 cm^3 (1.49 in^3) [9]. Keeping a constant width required a longitudinal spacing of .254 m (10 in). In terms of the transverse stiffeners, the optimized plate retains the same number of fixed and floating frames, and retains the same stiffener size for the floating frames. However, the fixed transverse frame size may be reduced to a 12.7 cm (5" x 3.7 lb/ft) extruded aluminum standard I-beam. The weight of the optimized panel is 294 kg (648 lb), resulting in a weight savings of approximately 15%.

Description	Value
Plating material	Aluminum 5086-H116
Stiffener material	Aluminum 5086-H111
Panel length	4.572 m (15 ft)
Panel width	1.829 m (6 ft)
Plate thickness	.635 cm (.25 in)
Longitudinal stiffeners	5 - 7.62 cm (3" x 1.64 lb/ft) Al I-beam

Span between longitudinal stiffeners	.254 m (10 in)
Span between transverse stiffeners	.762 m (30 in)
Transverse floating stiffeners	3 - 17.78 cm (7" x 5.8 lb/ft) Al I-beam
Transverse stiffeners	2 - 12.7 cm (5" x 3.7 lb/ft) Al I-beam

Table VI. Optimized Panel Geometry

While obtaining an optimum hull design based on weight is the objective, the cost to produce such a hull panel cannot be ignored. Therefore, a cost analysis that considers the change in cost to produce the initial design versus the optimum design was carried out. The variable cost required to produce the optimum design is written in terms of an incremental cost equivalent relative weight (iCERW) given by [10] as,

$$iCERW = \Delta material\ weight + K*\Delta man\text{-}hours,\ (kg)$$

where K is the ratio of the labor cost per hour to the cost per kilogram of aluminum [11]. In this case, a labor rate of \$50/hr was used [12] along with a material cost of \$4.40/kg. The additional longitudinal stiffener required for the optimum design, demands additional labor in terms of marking, positioning, aligning, fit and tack, and fillet welding along the stiffener length. An estimated two additional man-hours are required for this task. Given these estimates, the iCERW is -30.2 kg, indicating that the decrease in material weight offsets the increase in required man-hours.

The results indicate that optimization programs of this type can be a valuable tool that can be used at both the preliminary and contract design stage. Parametric studies performed through this study were essential in order to realize a cost effective lightweight aluminum hull structure.

Future enhancements will include stiffener and plate combinations that are evaluated in terms of both structural performance and overall cost. Other enhancements will include a sensitivity study of the various design and fabrication parameters on the overall cost.

CONCLUSIONS

This paper has presented the results of a design study for a cost effective, light weight, high strength aluminum hull panel. The hull panel was designed for panel line production and modular construction. It features the use of aluminum extrusions and alternating floating transverse frames to reduce production costs and minimize material weight.

In order to achieve these results, a 5 ft x 15 ft aluminum panel with alternate floating frames was tested in the UNO structural tester. The results were then compared with predictions made using the finite element method. The main conclusions of this study are:
1) the calculated deflection is slightly larger than the experimental measurements,
2) the calculated strains in the grillage are slightly lower than the

experimentally measured strains for the same loading, and

3) the calculated results show that the floating frame can meet the required loads.

The introduction of the HULLOPT procedure resulted in a systematic procedure to minimize the frame weight. This was accomplished by a parametric analysis based on available aluminum extrusions. The HULLOPT technique presented provides an effective method for optimizing the design of stiffened plates. The main conclusions of the optimization study are:

4) using the HULLOPT procedure and selection of available extrusions resulted in a 15% reduction in the panel weight, and

5) based on the incremental cost equivalent relative weight (iCERW), it can be shown that the reduction in weight offsets the increase in production man-hours, resulting in a net savings.

ACKNOWLEDGMENTS

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